Chapter 5

RADIANT COOLING IN US OFFICE BUILDINGS: RESULTS OF THE MODELING PROJECT

Chapter 4 described the parametric study designed for examining the topic of compatibility between office buildings equipped with radiant cooling systems and climates representative for the US. Because the study is based on RADCOOL simulations, and because RADCOOL has certain limitations, several assumptions were necessary regarding the base-case space to be modeled in the study (Table 4.4). To capture most of the characteristics of a wide range of US climates, a selection process allowed the choice of a small number of representative US locations. A different selection process was then employed to choose two location-specific week-long time periods for which the space simulation was carried. This chapter presents the results of the parametric study.

5.1 Chapter Outline

The indoor air temperature and humidity ratio of the space as simulated by RADCOOL and DOE-2 are presented in Section 5.2. The section focuses on the indoor conditions at the New Orleans, LA location. Common sense indicates that operating the radiant cooling system in this hot-humid Louisiana climate should be difficult: reducing the risk of condensation on the cooling surface represents a significant challenge. The section compares the indoor air temperature and relative humidity provided by the simulated radiant cooling and all-air systems, discusses the heat transfer phenomena specific to the two systems, and examines the effectiveness with which the night ventilation strategies studied reduce the risk of condensation on the cooling surface. Because the results obtained in the other 10 climates selected for the study are qualitatively similar, discussing the simulated space indoor conditions in all climates would be redundant.

Appendix B contains the results of the energy consumption and peak power demand calculations for the radiant cooling system and the all-air system conditioning the space located in the 11 climates of the study. Section 5.3 discusses the results for the radiant cooling system, while Section 5.4 discusses the results for the all-air system. In Section 5.5 the energy consumption and peak power demand of the radiant cooling system and all-air system are compared. Based on this comparison, the savings potential of the radiant cooling system is calculated and a quantitative relationship is derived linking the savings potential of the radiant cooling system with the "opportunity for savings" offered by the all-air system. To verify the applicability of this quantitative relationship for other building structures and other space orientations, Section 5.6 describes a few additional simulations. Section 5.7 summarizes the conclusions of the parametric study.

5.2 Indoor Conditions

The simulations conducted in the parametric study involve modeling of a base-case space with two ventilation strategies, at 11 representative US locations, during two week-long periods for each location. The base-case space simulated by the study is MBC2 in Figure 4.1. The facade of the space is exposed to climate-induced loads; its window has a south-western orientation. The lateral walls, ceiling and floor of the space are considered to be exposed to the same conditions on both of their surfaces. The "back wall" separates the space from a hallway with a constant air temperature. Table 4.4 summarizes the modeling assumptions of the parametric study.

The base-case space serves as a two-person office between 8 a.m. and 5 p.m., Monday through Friday, and contains some equipment generating heat during occupancy hours. To remove the sensible and latent loads due to indoor activity and solar gains, the space is conditioned by an air-conditioning terminal (the air supply register of an all-air system, or the radiant surface and air supply register of a radiant cooling system). The air-conditioning terminal maintains the indoor air temperature within one degree of the 24 °C setpoint, and the indoor air relative humidity below 60% during occupancy hours. Space cooling terminates at the end of the work day. The night ventilation strategy employed (see Figure 4.5 and Table 4.4) dictates whether or not dehumidified air is supplied to the space during off-occupancy hours. If space ventilation occurs through the night, the fresh air supplied is not only dehumidified but also cooled. The cooling power of this low volume flow of air is relatively small.

Figure 5.1 shows the simulated indoor air temperature of the base-case space during the week of system peak at the New Orleans, LA location, for the first ventilation strategy (space continuously ventilated, albeit half rate at night). As the peak demand occurs on Monday, July 25 at the New Orleans location, the week of system peak centered on this day is Friday, July 22, through Thursday, July 28 (i.e. the day of the all-air system peak demand is the fourth day in the figures). To illustrate the influence of mechanical cooling and ventilation on the indoor air temperature, Figure 5.2 shows the hourly variation of this temperature during the day of system peak. Figure 5.3 presents the simulated indoor air relative humidity during the week of system peak in New Orleans, for the space ventilated continuously. To facilitate a discussion of humidity control strategies for the radiant cooling system, Figure 5.4 compares the radiant surface temperature and dew-point temperature of the base-case space ventilated continuously. Figures 5.5 through 5.8 present results similar to those in Figures 5.1 through 5.4, but corresponding to the second ventilation strategy (space ventilation interrupted at night).

Results similar to those presented in Figures 5.1 through 5.8 were obtained for the week of system peak *and* the typical week at all 11 locations selected for the study.

Due to the design of the parametric study, the indoor air temperature and the indoor relative humidity during occupancy hours (8 a.m. to 5 p.m. on workdays) are similar for the
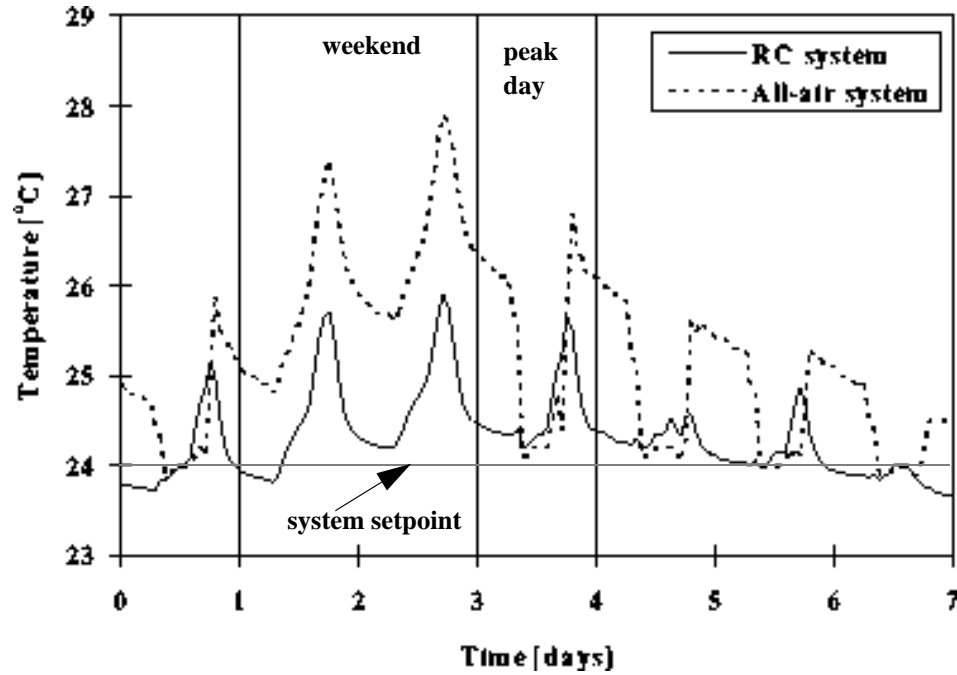
Figure 5.1. Indoor air temperature comparison at the New Orleans location during the week of system peak. Space ventilated continuously, half rate at night.
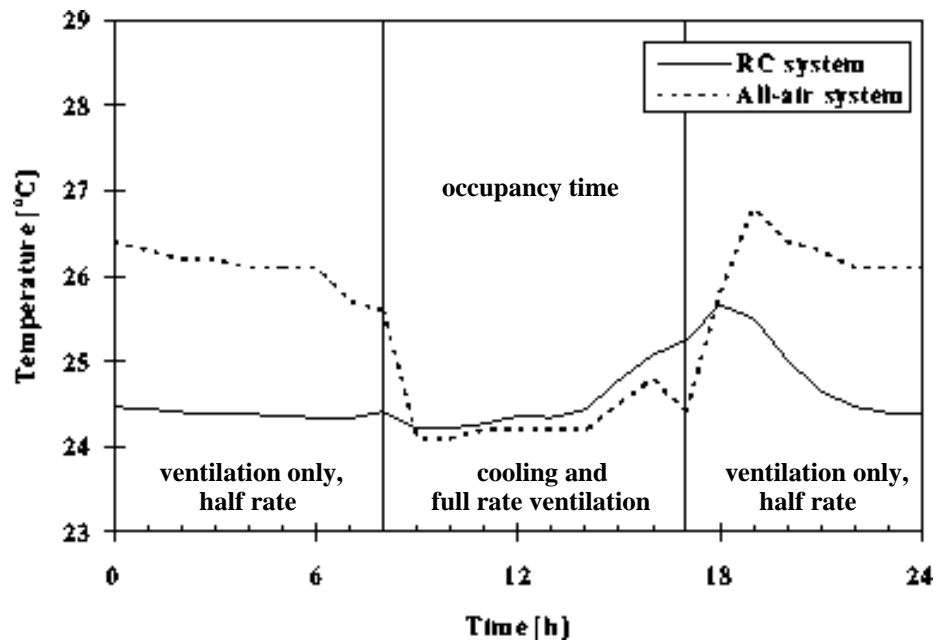

Figure 5.2. Indoor air temperature comparison at the New Orleans location during the day of system peak. Space ventilated continuously, half rate at night.
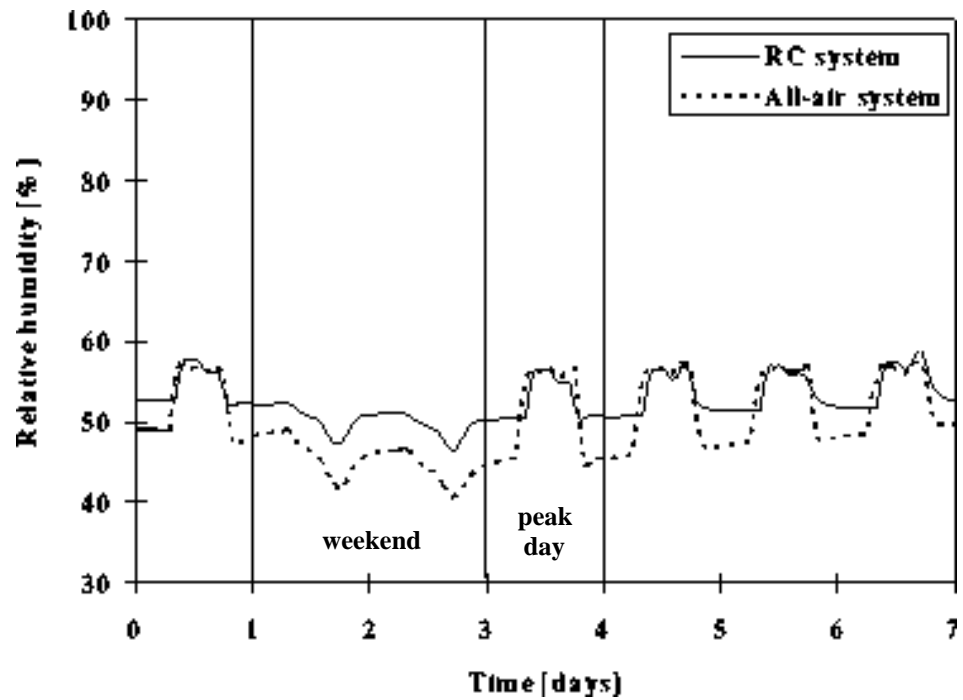
Figure 5.3. Indoor air relative humidity comparison at the New Orleans location during the week of system peak. Space ventilated continuously, half rate at night.
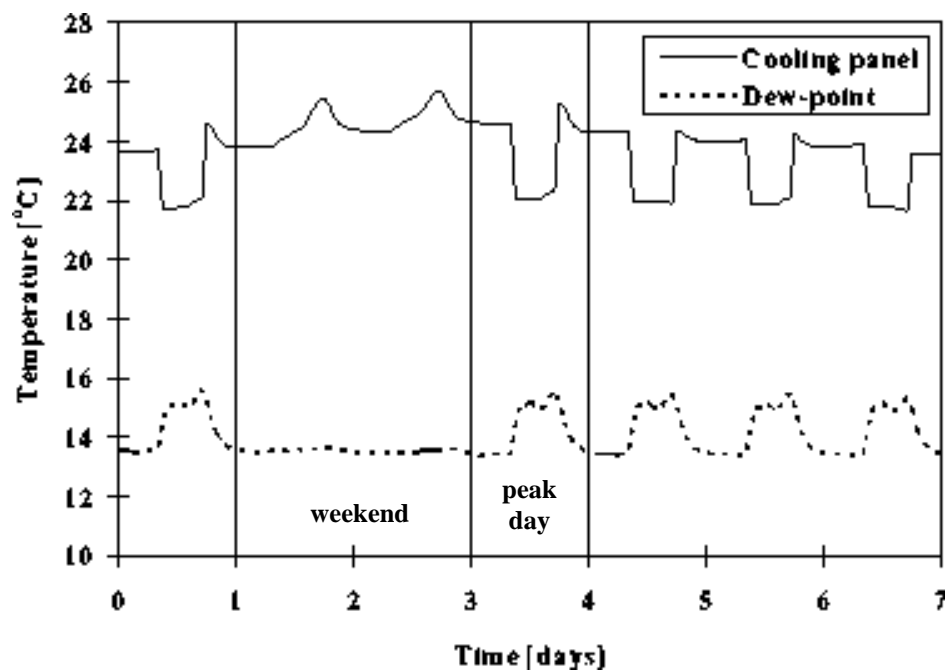

Figure 5.4. Comparison of cooling panel surface temperature and space dew-point temperature. New Orleans, space ventilated continuously, half rate at
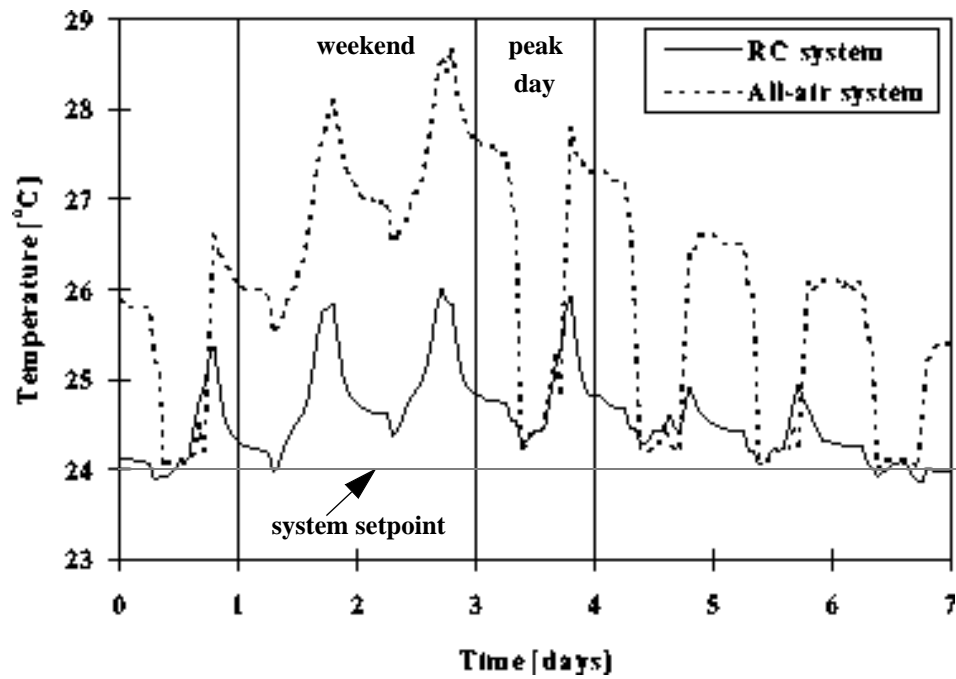
Figure 5.5. Indoor air temperature comparison at the New Orleans location during the week of system peak. Space ventilation interrupted at night.

Figure 5.6. Indoor air comparison at the New Orleans location during the day of system peak. Space ventilation interrupted at night.

Figure 5.7. Indoor air relative humidity comparison at the New Orleans location during the week of system peak. Space ventilation interrupted at night.
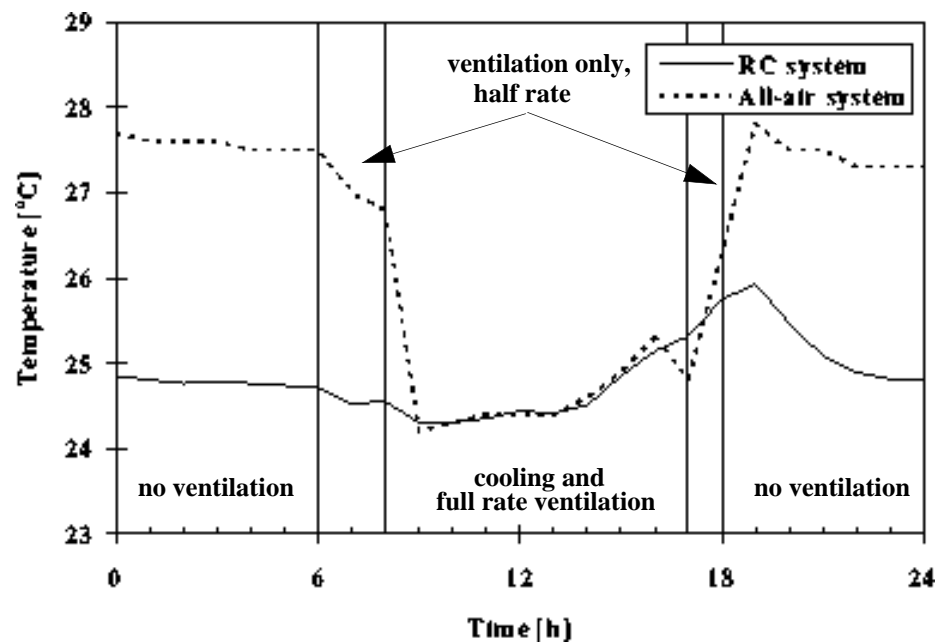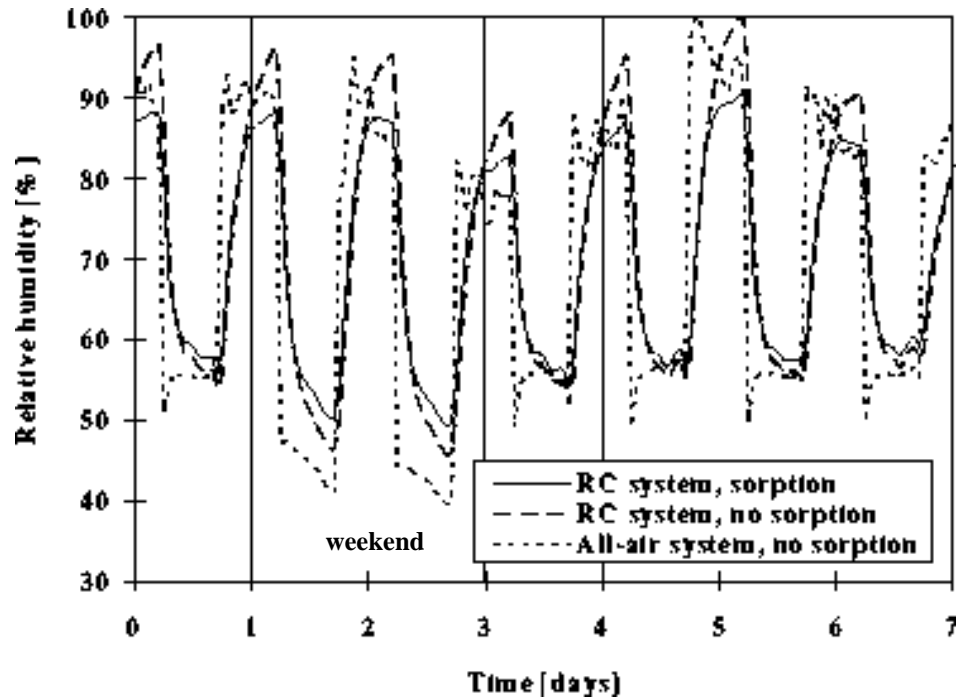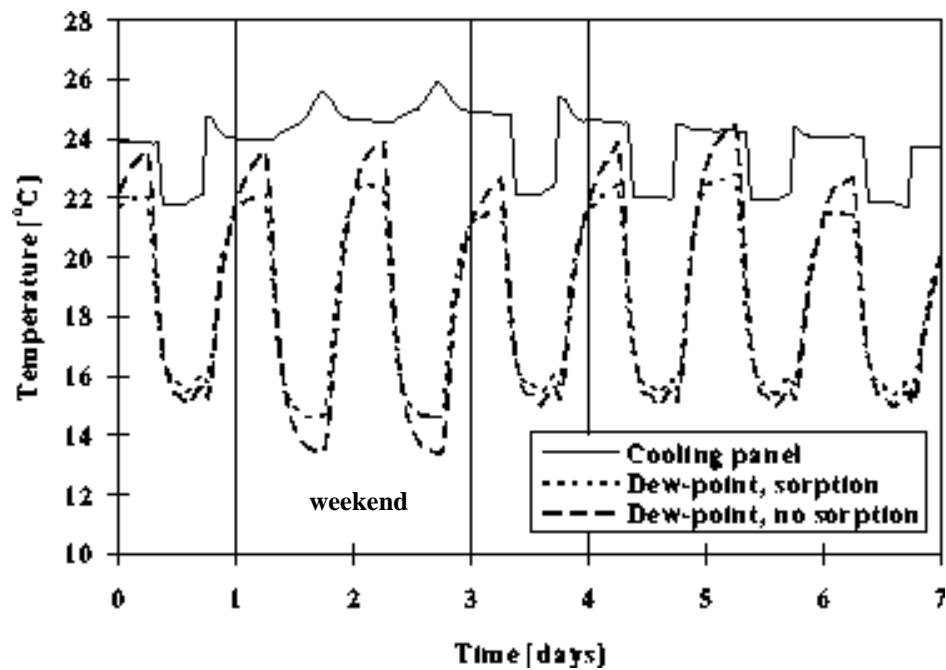

Figure 5.8. Comparison of cooling panel surface temperature and space dew-point temperature. New Orleans, space ventilation interrupted at night.

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space equipped with the radiant cooling system and the space equipped with the all-air system. The indoor air temperature presents a variation of a few tenths of a degree around the 24 °C setpoint (Figures 5.2 and 5.6), and the indoor relative humidity presents a variation of a few percent around the value of 60% (Figures 5.3 and 5.7). The indoor air temperature and relative humidity in the space conditioned by the radiant cooling system and the all-air system are not exactly the same because the two systems employ different mechanisms to provide space conditioning.

The indoor air temperature presents a higher variation during off-occupancy hours (Figures 5.2 and 5.6) and the two weekend days (Figures 5.1 and 5.5). This occurs because the main source of cooling for the space (the ceiling panels for the radiant cooling system, and the recirculated fraction of the supply air for the all-air system) is switched off during this time. Even if the space is ventilated during the night, the cooling power of this low volume flow is relatively small.

Figures 5.2 and 5.6 demonstrate that, for a 24-hour period, the indoor air temperature of the space conditioned by the radiant cooling system is more stable than that in the space conditioned by the all-air system. The mechanisms used by the two systems to cool the space explain this result. The heat removal mechanism employed by the all-air system provides cooling to the indoor air directly, and to the occupants indirectly, through convective exchange with the cool air. Because convective heat exchange does not cool the surfaces of the space very efficiently, they store heat during the day.[1] When the mechanical cooling stops at the end of occupancy hours, the space surfaces release the stored heat, causing a sharp increase of the indoor air temperature. The amplitude of this increase is 2.5 to 3 °C, higher if space ventilation also stops. Then the space cools slightly overnight. When the space ventilation starts before the next occupancy period, the indoor air temperature presents a slow decrease. A rapid 1 - 2 °C decrease follows after the mechanical cooling is switched on.

By comparison, the radiative heat exchange mechanism employed by the radiant cooling system provides cooling to the occupants directly through radiation, and to the indoor air indirectly through convective heat exchange with the cooled ceiling. Because the vertical walls and the floor also exchange heat with the cooled ceiling, they are actively cooled during the day, therefore they can store less heat than their counterparts in the space conditioned by the all-air system. It is important to note that although they are actively cooled during the day, the vertical walls and the floor are still warmer than the indoor air. When the cooling stops, these building components release the heat stored during the day. Consequently, the indoor air temperature increases sharply, but only by

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1. The assumption used to model the lateral walls, ceiling and floor is that these surfaces have the same boundaries on both surfaces (temperature and heat flux). While this translates into zero heat transfer through the midpoint of such a building component, it does not prevent each half of the building component from storing and releasing heat into the space to which it belongs.

1.5 - 2 °C. The indoor air temperature of the space conditioned by the radiant cooling system is thus more stable over a 24-hour period than the indoor air temperature of the space conditioned by the all-air system.

The first ventilation strategy supplies dehumidified air continuously to the base-case space to pressurize the space and avoid humidity buildup by infiltration. Figure 5.3 demonstrates that this strategy maintains the relative humidity of the space inside the comfort range at all times.[1] To save dehumidification and fan energy during off-occupancy hours, the second ventilation strategy interrupts the ventilation of the base-case space for 12 hours overnight. The moisture mass balance performed in this case shows that, due to infiltration-driven humidity buildup, the relative humidity of the indoor air increases substantially during the 12 hours when the space is not pressurized (see Figure 5.7).

Figures 5.4 and 5.8 demonstrate that dehumidifying the supply air to maintain the indoor air relative humidity just below the upper limit of the comfort range (60%) during occupancy hours lowers the dew-point temperature inside the space to about 15.5 °C. If cooling water at 20 °C is supplied to the ceiling panels during this time, the mean temperature of the radiant surface becomes about 22 °C. A temperature difference of 6.5 °C between the average radiant surface temperature and the dew-point temperature (4.5 °C between the coldest end of the ceiling panes and the dew-point temperature) is adequate to ensure that no condensation forms on the surface of the ceiling panels.

After the supply of cooling water has been discontinued, radiation from the vertical walls and floor causes the panel surface temperature to increase to around 24 °C. Assuming that infiltration with moist outside air can be avoided by supplying ventilation air at half rate (Figure 5.4), dehumidifying the ventilation air to 9.5 g water/kg dry air maintains the dew-point temperature of the space around 13.5 °C during off-occupancy hours. The 8.5 °C temperature difference between the panel surface and the dew-point indicates that condensation does not form on the surface of the ceiling panels during off-occupancy hours. Dehumidifying the outside air to about 15.5 g water/kg of dry air would have been sufficient to maintain this temperature difference at 3 °C.

If ventilation with dehumidified air is discontinued during off-occupancy hours, or if infiltration with moist outside air cannot be avoided, condensation may appear on the surface of the ceiling panels. The data presented in Figure 5.7 and 5.8 show that it is important to account for the mechanism of moisture sorption on the space surfaces (see Appendix A) when examining the effects of infiltration on the moisture balance of the indoor air.

Specifically, if the moisture balance does not account for sorption (as in the case of DOE-2, for instance), the simulated indoor air relative humidity becomes equal to the

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1. According to ASHRAE Standard 55-1992 [1] the comfort range for the relative humidity in an office space setting (light sedentary activity, and occupants wearing clothing adequate to the season) is 30 to 60%.

outside air relative humidity shortly after space ventilation is interrupted. Because none of the moisture migrating into the space is stored in the surfaces, the simulation results for the week of system peak in New Orleans indicate that the indoor air reaches saturation in the early morning hours of the fifth day (dotted line for DOE-2 results and dashed line for RADCOOL results in Figure 5.7). As the ceiling panels are colder than the indoor air, air saturation indicates the presence of condensation on the surface of the panels. Figure 5.8 confirms that during the early morning hours of the fifth day the dew-point temperature of the space becomes higher than the surface temperature of the ceiling panels.

If sorption is accounted for in the moisture mass balance,[1] the simulation results show that the relative humidity of the indoor air increases during the off-occupancy hours, but the air does not become saturated (solid line for RADCOOL results in Figure 5.7). Figure 5.8 confirms that in this case, the dew-point temperature inside the space remains at least one degree lower than the temperature of the radiant surface.

As Figures 5.1 and 5.5 show, supplying cold water at 20 °C to the ceiling panels at the New Orleans location provides the radiant surface with sufficient cooling power to remove the cooling loads from the base-case space. If the cooling water were supplied at a temperature lower than 20 °C (to increase the cooling power of the radiant surface, for example), the ceiling panels would not warm up past the dew-point temperature of the space during the off-occupancy hours, and condensation would form on the panel surface. Consequently, if the supply water temperature at the New Orleans location were lower than 20 °C, space ventilation with dehumidified air would be strongly recommended to avoid condensation on the panel surface. Similar results were obtained for the Cape Hatteras, New York, Fort Worth, and Chicago locations. At the other 6 locations, infiltration with outside air does not increase the dew-point temperature of the space past 18 °C, therefore lowering the water supply temperature by a few degrees does not increase the risk of condensation.

If the space could not be pressurized and/or infiltration could not be avoided, supplying dehumidified air during off-occupancy hours would reduce the relative humidity inside the space and would reduce the risk of condensation. The optimum level of dehumidification of the supply air in such a case is subject for future research.

It is important to note that moisture sorption on the walls has a significant influence on the moisture mass balance of the indoor air only when the relative humidity of the air presents a large variation (for example, when moisture is produced or transported inside the space).[2] In such a case, some of the moisture is stored in the walls by sorption, and

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1. The RADCOOL calculation assumes that the space walls are covered in an oil-based paint and the floor is covered in linoleum. Since the radiant surface modeled consists of aluminum panels, no sorption is modeled for the ceiling panels.

2. Here the term "wall" refers to any space surface that is not covered with cooling panels.

the indoor air relative humidity increases at a slower rate. Later, when the source of moisture has disappeared, the walls dry out, thus releasing the moisture back into the air. Conversely, when the indoor air relative humidity varies very little over time, as in the case when the space is continuously ventilated (Figure 5.3), there is no significant moisture sorption in the walls. Consequently, when the space is pressurized to avoid infiltration, sorption can be safely ignored when performing the moisture mass balance for the indoor air.

It is also important to note that the radiant cooling system can maintain the indoor air temperature around the setpoint of 24 °C because (1) the structure of the space is well insulated, and (2) the internal loads are relatively low. Because the radiant cooling systems currently available on the market have a maximum cooling power of 140 W/m^2, these systems might not be able to supply sufficient cooling to a poorly-insulated space with high internal loads. Common sense indicates that the boundary between the domains in which radiant cooling systems might and might not supply sufficient cooling is also a function of climate. Identifying this boundary is subject to further research.

5.3 The Energy Consumption and Peak Power Demand of the Radiant Cooling System

The parameters used in the RADCOOL simulations of the space conditioned by the radiant cooling system, and in the DOE-2 simulations of the space conditioned by the all-air system, allow the calculation of the sensible, latent and distribution loads for each system terminal. This section discusses the results of the calculation performed for the radiant cooling system while the next section discusses the results for the all-air system.

According to the results of the parametric study, at 9 of the 11 locations examined, supplying cooling water at 20 °C to the ceiling panels allows the radiant cooling system to maintain the indoor air temperature within one degree of the 24 °C design setpoint during occupancy hours. If the moisture mass balance accounts for sorption on the walls and floor of the space, the simulation results indicate that condensation does not form on the surface of the ceiling panels at any of the locations studied. This statement holds for the typical week and the week of system peak, and for both ventilation strategies.

The locations where the ceiling panels do not have sufficient cooling power if the cooling water is supplied at 20 °C are Phoenix, AZ and Salt Lake City, UT. In these two climates the daily maximum radiant load exceeds 40 W/m^2, the outside air temperature exceeds 35 °C, and the outside relative humidity is 10% on average (during both the typical week and the week of system peak). At these locations the radiant cooling system cannot maintain the ambient temperature near the 24 °C setpoint unless the cooling water and ventilation air are supplied at 17.5 °C. Because Phoenix and Salt Lake City are dry locations, lowering the supply water temperature does not increase the risk of

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condensation on the ceiling panels, and lowering the temperature of the supply air does not increase the latent load of the system.

5.3.1 Energy consumption of the radiant cooling system

When the space is ventilated continuously, night ventilation contributes to the total energy consumption of the radiant cooling system terminal. Night ventilation accounts for a larger fraction of the total energy consumption of the system terminal in hot, humid climates where air dehumidification is energy-intensive. In cooler and drier climates, night ventilation contributes only marginally to the total all-air system energy consumption. Night ventilation accounts for a fraction between 2% (Seattle) and 19% (New Orleans) of the total energy consumption of the radiant cooling system terminal during the typical week, and between 4% (Seattle) and 26% (New Orleans) during the week of system peak.

At all locations studied, the energy consumption due to cooling and dehumidifying the ventilation air is *lower* when the space ventilation is interrupted at night (second ventilation strategy) than when the space is ventilated continuously (first ventilation strategy). The decrease in energy consumption due to interrupting the space ventilation at night is greater in moist climates, where the outside air often becomes saturated at night. Interrupting the space ventilation at night avoids the dehumidification energy consumption associated with conditioning this very moist air.

In the parametric study, the main source of space cooling is switched off at the end of the occupancy hours. In the case of the radiant cooling system terminal, this translates into interrupting the supply of cooling water to the ceiling panels. This reduces the radiative cooling of the other surfaces significantly, but not entirely, as the ceiling panels are still colder than the other surfaces. Interrupting the space ventilation an hour later (second ventilation strategy) eliminates the forced convective cooling of the space as well. Consequently, when the vertical walls and floor release the heat accumulated during the day, the cooler ceiling absorbs a higher quantity of heat than it would absorb if the space were still ventilated. When the cooling water supply to the ceiling panels is switched on again the next day, the water must cool the warmer ceiling surface before the ceiling itself can cool the space. Therefore, the cooling coil energy consumption due to water cooling *increases* when the space ventilation is interrupted at night. This increase happens at all locations, and is highest in hot dry climates.

When space ventilation is interrupted at night, the avoided sensible and latent cooling coil energy consumption prevails over the increase in the sensible cooling coil energy consumption for water cooling. Consequently, the total energy consumption of the radiant cooling system *decreases* when the ventilation is interrupted at night. The reduction in total energy consumption is in the range from 2% (New York) to 18% (New Orleans) during the typical week, and from 4% (Seattle) to 26% (New Orleans) during the week

of system peak. The reduction is higher during the week of system peak because the energy benefits associated with interrupting the space ventilation at night are larger in the hot season. The reduction is highest in hot humid climates.

At the level of the cooling coil serving the radiant cooling system, the energy consumption due to water sensible cooling is higher than the energy consumption for air sensible cooling at all locations studied. This is consistent with the fact that, by design, the radiant cooling system cools the space mainly by radiation, and water is the cooling agent connecting the radiant surface to the cooling coil. When the space is continuously ventilated, water cooling accounts for a fraction between 70% (New Orleans) and 98% (Seattle) of the cooling coil sensible energy consumption during the typical week, and for a fraction between 59% (Phoenix) and 87% (Seattle) during the week of system peak. When space ventilation is interrupted at night, the energy consumption due to air sensible cooling decreases and the energy consumption due to water cooling increases. Water cooling accounts for a fraction between 75% (New Orleans) and 98% (Seattle) of the cooling coil sensible energy consumption during the typical week, and for a fraction between 70% (New Orleans) and 89% (Seattle) during the week of system peak.

The energy consumption due to air dehumidification varies widely across the climates. When the space is continuously ventilated, the latent fraction of the total energy consumption of the cooling coil is in the range from 0% (Salt Lake City) to 41% (New Orleans) during the typical week, and from 0% (Salt Lake City) to 53% (New Orleans) during the week of system peak. When the space ventilation is interrupted at night, the energy consumption due to dehumidification decreases. The latent fraction of the total energy consumption of the cooling coil is in the range from 0% (Salt Lake City) to 32% (New Orleans) during the typical week, and from 0% (Salt Lake City) to 43% (Cape Hatteras) during the week of system peak.

Because the radiant cooling system supplies the same (constant) air and water volumes to the system terminal at all locations studied, the energy consumption due to water distribution (pump) and air distribution (fan) are the same at all locations, during the typical week and the week of system peak.

5.3.2 Peak power demand of the radiant cooling system

The peak power demand due to conditioning the space does not vary much across the climates.[1] When the space is continuously ventilated, the peak electrical power demand of the radiant cooling system is in the range from 20.5 W_e/m^2 (Seattle) to 30.3 W_e/m^2 (Cape Hatteras). When the ventilation is interrupted at night, the space is not mechani-

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1. The total (hourly) load due to space conditioning is calculated as the sum between the sensible and latent loads on the cooling coil (due to air cooling and dehumidification and water cooling), and the fan and pump loads (due to air and water distribution). The peak power demand is the highest hourly load.

cally cooled, and the peak electrical power demand increases. In this case, the electrical peak power demand of the radiant cooling system varies in the range from 20.9 W_e/m^2 (Seattle) to 30.7 W_e/m^2 (Cape Hatteras).

As discussed in Section 4.5.1, the load calculations assumed that the cooling coil-chiller combination serving the radiant cooling system terminal has a constant coefficient of performance, COP = 3. Because the fan and pump load contributions to the total peak power demand are small, the ratio of the peak thermal load to the peak electrical load is almost 3:1. The results reported in this section imply that the radiant cooling system can successfully remove a thermal load of about 60 W/m^2 (Seattle) - 90 W/m^2 (Cape Hatteras) from the base-case space. This range is consistent with the cooling power of radiant cooling systems currently available on the market [2].

In what follows, the calculations for the energy consumption and peak power demand of the all-air system also assume a constant COP = 3 for its cooling coil-chiller combination. It is important to note, however, that both the chiller serving the radiant cooling system and the chiller serving the all-air system function in part-load during most of their on-time. The COP of a chiller in part-load is lower than the COP at design point.

5.4 The Energy Consumption and Peak Power Demand of the All-Air System

5.4.1 Energy consumption of the all-air system

The all-air system employs a variable air volume system during occupancy hours (8 a.m. to 5 p.m.) and a constant volume system, or no system at all, during off-occupancy hours. When the space is ventilated continuously, night ventilation contributes to the total energy consumption of the all-air system terminal. Night ventilation accounts for a fraction of 2% (Seattle) to 19% (Fort Worth) of the total energy consumption during the typical week, and for a fraction of 3% (Seattle) to 23% (New Orleans) during the week of system peak.

Depending on the outside air conditions, interrupting the space ventilation at night sometimes leads to a decrease, other times to an increase in the cooling coil *sensible* energy consumption (due to cooling the supply air). Two factors contribute to this result. First, if the space is continuously ventilated, some of the energy stored in the walls during the day is removed at night. Interrupting the space ventilation at night reduces heat removal and leads to an increase of the cooling coil sensible load during the next day. Second, the cooling coil on-time is longer when space is ventilated continuously than when the space ventilation is interrupted at night.

Depending which of the two factors prevails, the cooling coil sensible energy consumption will increase or decrease when the space ventilation is interrupted. The following rule of thumb holds for the climates selected for the parametric study: if the daily mini-

mum of the outside air temperature is higher than 18 °C, the continuous ventilation of the space is associated with a high cooling coil energy consumption. Consequently, the energy saved by reducing the on-time of the cooling coil offsets the extra energy that must be removed from the space the next day due to heat not released from the building structure. Overall, at locations where the daily minimum outside air temperature is higher than 18 °C, the cooling coil sensible energy consumption *decreases* when the space ventilation is interrupted at night. These locations are New Orleans, Fort Worth, and Phoenix during the typical week, and New Orleans, Cape Hatteras, New York, Fort Worth, Chicago, Boston, Phoenix, and Salt Lake City during the week of system peak.

If the daily minimum is below 18 °C, the energy required to cool the night ventilation air is minimal. Consequently, when the space ventilation is interrupted at night, the extra energy that must be removed from the space the next day offsets the savings achieved by reducing the cooling coil on-time. Thus the cooling coil sensible energy consumption *increases* if space ventilation is interrupted at night at the locations where the daily minimum of the outside air temperature is below 18 °C. This happens in New York, Chicago, Boston, San Jose, Scottsbluff, Salt Lake City, and Seattle during the typical week, and in San Jose, Scottsbluff, and Seattle during the week of system peak.

Similarly to the cooling coil serving the radiant cooling system terminal, the energy necessary for air dehumidification by the cooling coil serving the all-air system terminal varies widely across the climates. When the space is continuously ventilated, the latent fraction of the cooling coil total energy consumption is in the range from 0% (Salt Lake City) to 30% (New Orleans) during the typical week, and from 0% (Salt Lake City) to 40% (New Orleans) during the week of system peak. When space ventilation is interrupted at night, the energy consumption due to latent heat removal decreases at all locations where dehumidification is required. The reduction in dehumidification energy is higher in the hot climates. The latent fraction of the total cooling coil energy consumption varies in the range from 0% (Salt Lake City) to 22% (New Orleans) during the typical week, and from 0% (Salt Lake City) to 28% (New Orleans) during the week of system peak.

At all locations, the all-air system must remove a larger quantity of energy from the space when the space ventilation is interrupted at night than when the space is continuously ventilated. This additional energy has a sensible component, mostly due to higher heat release from the walls into the space in the absence of ventilation, and a latent component, mostly due to humidity buildup through infiltration in the absence of ventilation. The all-air system removes the additional energy the next day. This leads to an increase in cooling coil energy consumption, but also to an increase in fan energy consumption (the larger air volume supplied to meet the higher load requires a larger fan). The increase in fan energy consumption due to the interruption of space ventilation at night is larger in the moist climates, where moisture buildup is large. The range of the increase in fan energy consumption due to interrupting the space ventilation at night is from 9% (Salt Lake City) to 48% (New York) during the typical week, and from 8% (Salt Lake

City) to 39% (New York) during the week of system peak.

5.4.2 Peak power demand of the all-air system

As in the case of the radiant cooling system terminal, the total peak power demand of the all-air system terminal does not vary much across building locations. When the space is continuously ventilated, the electrical peak power demand of the system is in the range from 29.8 W_e/m^2 (Seattle) to 45.9 W_e/m^2 (Phoenix). The peak power demand of the system is higher for the all-air system than for the radiant cooling system mainly due to the larger fan power demand of the all-air system. When the space ventilation is interrupted at night, the electrical peak power demand of the all-air system increases, and is in the range from 30.3 W_e/m^2 (Seattle) to 48.5 W_e/m^2 (Phoenix).

5.5 Comparison of the Performance of the Radiant Cooling System and of the All-Air System

5.5.1 Energy consumption

At all the locations studied the energy consumption of the radiant cooling system terminal was lower than the energy consumption of the all-air system terminal. This statement holds for the typical week as well as for the week of system peak, and for both ventilation strategies. The purpose of this Section is to quantify the energy savings.

The results of the parametric study show that the numerical value of the energy savings achieved by replacing the all-air system terminal with the radiant cooling system terminal varies as a function of the building location. Savings in moist climates are lower than savings in dry climates. This result follows from one of the assumptions of the parametric study, namely that both systems condition (cool and dehumidify) the same quantity of outside air.[1] Because the radiant cooling system and the all-air system maintain a similar relative humidity inside the space, the energy required for dehumidification is similar for the two systems, and the dehumidification process does not provide any "opportunity for savings". The "opportunity for savings" resides in the fact that removing heat from the space by circulating relatively large volumes of air is more energy-intensive than removing heat from the space by circulating water and ventilation air. In other words, the sensible air cooling and fan energy consumption of the all-air system are higher than the sensible air cooling, sensible water cooling, fan and pump energy consumption of the radiant cooling system. The sensible and fan energy provide less "opportunity for savings" in moist climates than in dry climates because in moist cli-

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1. In an ideal situation, the operation of the radiant cooling system and that of the all-air system would be matched based on the indoor effective temperature of the base-case space. It is unclear whether the results reported in this section are applicable to such a case.

mates dehumidification accounts for a large fraction of the total energy consumption.

At all locations, the energy savings achieved when the space ventilation is interrupted at night are larger than the energy savings achieved when the space is ventilated continuously. This happens primarily because the all-air system provides an "opportunity for savings" mainly during occupancy hours.[1] Interrupting the space ventilation at night is associated for both systems with the need for more sensible cooling energy, and for the all-air system with the need of more fan energy during the next day. This increases the "opportunity for energy savings" when the ventilation is interrupted at night, as compared to the case when the space is continuously ventilated.

The results of the parametric study show that energy savings achievable during the typical week are different from the energy savings achievable during the week of system peak. The energy savings achieved during the week of system peak are higher than the savings achieved during the typical week at 7 of the 11 building locations, and lower than the savings achieved during the typical week at 4 of the 11 building locations.

The design and operation of the VAV system explain this result. In order to meet the cooling load at a given time, the VAV system adjusts the flow rate (and the temperature, if necessary) of the supply air. The supply air flow rate and temperature at each time step are thus a function of the temperature and moisture of the outside air, the temperature of the return air, and system design requirements such as the setpoints for the minimum supply air temperature and the indoor relative humidity. If the outside air is hot and moist, the cooling coil serving the VAV system must cool and dehumidify a relatively warm and moist air mix (between the required quantity of fresh air and recirculation air). In this situation the energy consumption of the cooling coil is high. If the outside air temperature is low, the cooling coil must cool and dehumidify a relatively cold and dry air mix. In this situation the energy consumption of the cooling coil serving the VAV system is low.

In the context of the parametric study, when the outside air temperature is lower during the typical week than during the week of system peak, the energy consumption of the all-air system during the typical week is lower than during the week of system peak. If the space is ventilated continuously, the energy consumption of the all-air system is further reduced during the typical week because the outside air is cold, therefore the cooling coil energy consumption is minimal.[2] By comparison, the radiant cooling system

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1. When the cooling has stopped at the end of occupancy hours, the radiant cooling system terminal and the all-air system terminal employ identical constant volume systems for ventilation. If the first ventilation strategy is employed, the relatively low cooling power of the ventilation air offers little "opportunity for savings". If the second ventilation strategy is employed, no ventilation air is supplied to the space, therefore there is no "opportunity for savings".

2. The ventilation system can supply air to the space at a minimum temperature dictated by a pre-set minimum surface temperature of the cooling coil (usually 15 °C for the all-air system in this study).

always supplies the same quantity of outside air at 20 °C (17.5 °C in Phoenix and Salt Lake City) and 9.5 g water/kg dry air, to minimize the risk of condensation on the cooling surface.[1] The energy consumption of the radiant cooling system also becomes lower during the typical week as compared to the week of system peak. However, the reduction in the energy consumption of the radiant cooling system is lower than that of the all-air system because (1) the space is mainly cooled by radiation during the day, and (2) at night the ventilation air is still supplied at 20 °C (or 17.5 °C) and 9.5 g water/kg dry air. Consequently, if the outside air temperature is lower during the typical week than during the week of system peak, the energy savings during the typical week are lower than the energy savings during the week of system peak. This happens in New York, Fort Worth, Boston, San Jose, Scottsbluff, Salt Lake City, and Seattle.

If the outside air temperature is high during the typical week (New Orleans, Cape Hatteras, Phoenix), or if the outside air moisture content is high during the typical week (Chicago), the all-air system functions at a point of relatively high energy consumption. The energy consumption is relatively high because (1) the cooling coil serving the all-air system must cool and dehumidify a mix of warm and/or moist outside and recirculation air, and (2) the air volume supplied to the space is relatively large, and so is the fan that circulates this air volume. By comparison, the energy consumption of the radiant cooling system is relatively low during the typical week because (1) this system removes most of the cooling loads by radiation; the connection between the radiant surface and the cooling coil is accomplished by water circulation, and (2) the quantity of outside air that the system must cool and dehumidify is much smaller than the quantity of mixed air that the all-air system must cool and dehumidify. Consequently, if the outside air temperature and/or the outside air humidity is high during the typical week, the energy savings during the typical week are higher than the energy savings during the week of system peak.

To summarize, the all-air system is favored in the climates where the weather during the typical week is cool and dry. Conversely, the radiant cooling system is favored in the climates where the weather during the typical week is warm and/or moist. However, at all locations, the energy consumption of the radiant cooling system is lower than the energy consumption of the all-air system. The potential energy savings during the typical week are in the range from 6% (Seattle) to 36% (Phoenix) when the space is ventilated continuously, and in the range from 23% (Seattle) to 42% (Phoenix) when the space ventilation is interrupted at night.[2] The average and standard deviation of the energy savings

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1. This does not imply that the radiant cooling system must mechanically *heat* the outside air at night. If the outside air temperature is less than 20 °C, it can be warmed up by using waste heat from the compressor operating the cooling coil, or by channeling it through building components (however, this strategy requires special design for the air inlet).

2. The energy savings are calculated as the difference between the total energy consumption of the all-air system and the total energy consumption of the radiant cooling system, divided by the total energy consumption of the all-air system.

during the typical week are 25.4% and 9.6% when the space is continuously ventilated, and 34.8% and 6.7% when the space ventilation is interrupted at night.

It important to note that, when the space is continuously ventilated, supplying fresh air at a temperature lower than 20 °C or (17.5 °C) in dry climates would *not* lead to condensation on the radiant surface. Consequently, if the radiant cooling system had been designed to take advantage of this opportunity to reduce the load on the cooling coil at night, the calculated energy savings would have been higher than those reported in this section.

5.5.2 Peak power demand

Due to the difference in heat removal mechanisms of the radiant cooling system and all-air system, the two systems reach their peak power demand at different times during the peak day. The time of peak of the all-air system usually happens shortly after noon. The time of peak of the radiant cooling system usually happens one or two hours later.

In all the climates studied the peak power demand of the radiant cooling system is lower than that of the all-air system. This statement is true for the typical week as well as for the week of system peak, and for both ventilation strategies. It can be explained based on (1) the heat removal mechanisms of the two systems (radiant vs. convective), and (2) the size of the fan employed by each of the two systems at the time of the peak demand (the radiant cooling system employs a much smaller fan than the all-air system).

The peak power savings do not vary much with the building location.[1] As in the case of the energy savings, the peak power savings are larger when the space ventilation is interrupted at night than when the space is continuously ventilated. This happens primarily because, when space ventilation is interrupted at night, the energy that must be removed during the next day increases, so the peak cooling demand increases for both systems. Because the all-air system cools the space mainly by convection, and because it employs a larger fan than the radiant cooling system, the increase in the peak power demand of the all-air system is larger than the increase in the peak power demand of the radiant cooling system.

At 9 of the 11 locations selected for the study the peak power savings during the typical week are *higher* than the peak power savings during the week of system peak. The locations where the peak power savings during the typical week are *lower* than the peak power savings during the week of system peak are New York and Boston. This result can be explained based on the weather conditions at the time of the peak load: during the

1. The peak power savings are calculated as the difference between the peak power demand of the all-air system and the peak power demand of the radiant cooling system, divided by the peak power demand of the all-air system.

week of system peak the weather is sunny, while during the typical week it is overcast, and/or raining. Consequently, the solar heat gain is high during the week of system peak and low during the typical week. Since the internal loads are assumed to not change during the year, the time and amplitude of the peak load are driven by the weather-induced loads. During the week of system peak in New York and Boston, the dominant component of the peak load is the sensible load due to space solar heat gain. During the typical week, the dominant component of the peak load is the latent load due to air dehumidification. As the cooling coils serving the radiant cooling system and the all-air system handle the same latent loads, the "opportunity for power savings" of the radiant cooling system is larger when the load is mainly sensible, and smaller when the load is mainly latent. Consequently, the peak power savings in New York and Boston are larger during the week of system peak than during the typical week.

The peak power savings during the week of system peak vary between 22% (New York) - 35% (Phoenix) when the space is continuously ventilated, and between 23% (New York) - 37% (Phoenix) when the space ventilation is interrupted at night. The average and standard deviation of the peak power savings over all the climates considered are 27.2% and 4.0% when the space is continuously ventilated, and 28.4% and 4.3% when the space ventilation is interrupted at night.

5.5.3 Climate-induced trends into the energy consumption and peak power savings of the radiant cooling system

The results reported in the preceding sections associate numerical values to the ability of the simulated radiant cooling system to save energy and peak power at a given location. These results can be presented in the form of a distribution of the energy and peak power savings with the number of locations at which given savings are achieved. Figure 5.9 shows the distribution corresponding to the second ventilation strategy. The results in Figure 5.9 show that, when the space ventilation is interrupted at night, the simulated radiant cooling system requires on average 35% less energy, and 28% less peak power than the simulated all-air system to provide similar indoor temperature and relative humidity to the base-case space during occupancy hours.

The distribution in Figure 5.9 does not provide the capability to predict the savings that could be achieved by replacing the all-air system with a radiant cooling system at a given location. At present, much information is available regarding the design and functioning of all-air systems, but little information is available regarding the design and functioning of radiant cooling systems. Consequently, a quantitative link between the energy and peak power savings of the simulated radiant cooling system and the energy consumption and peak power demand of the simulated all-air system would constitute a useful addition to the existing knowledge about radiant cooling systems. Such a link would also provide the means to estimate the savings that could be achieved if the radiant cooling system replaced the all-air system at any given location.
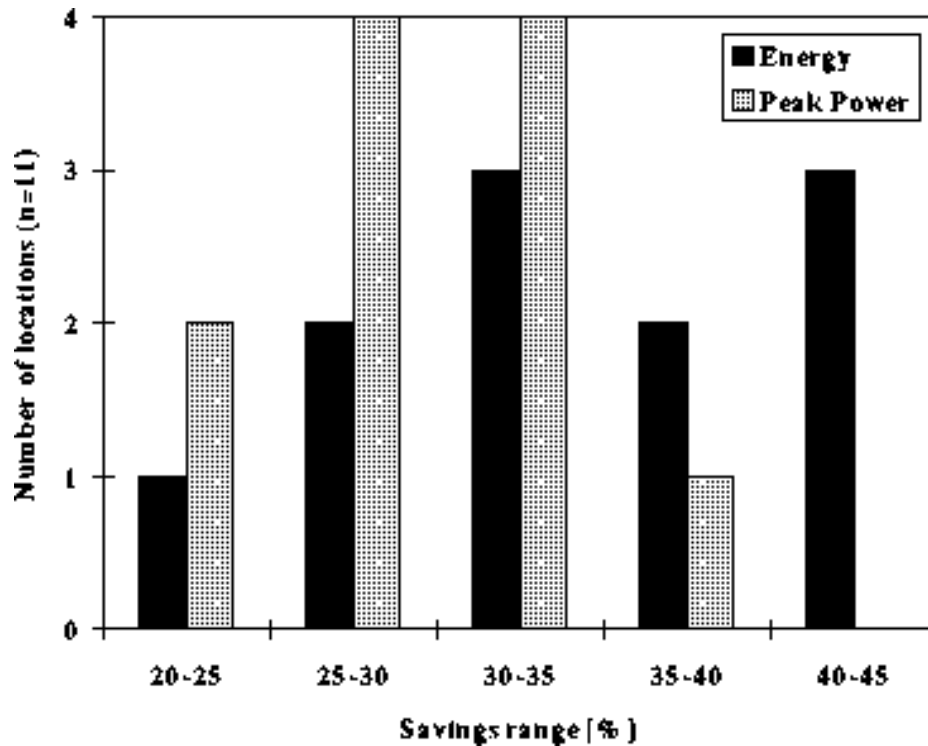
Figure 5.9. Distribution of the energy and peak power savings of the radiant cooling system with the number of locations. Space ventilation interrupted at night. Energy average = 34.8%, standard deviation = 6.7%. Peak power average = 28.4%, standard deviation = 4.3%.

The following observations were useful when establishing this quantitative link:

(1) The results of the parametric study show that the indoor conditions of the base-case space do not comply with the new version ASHRAE Standard 62 (currently under revision) for all cases studied. According to ASHRAE Standard 62R [3], the indoor air relative humidity should be maintained below 70% at all times. The results of the parametric study indicate that at the humid locations (New Orleans, Cape Hatteras, New York, Fort Worth, and Chicago), the indoor air relative humidity exceeds 70% if the space ventilation is interrupted at night. However, the indoor air relative humidity is always maintained below 70% at these locations if the space is ventilated continuously. To comply with ASHRAE Standard 62R, the base-case space located at the humid locations should therefore be continuously ventilated (should employ the first ventilation strategy).

(2) Interrupting the space ventilation at night at the drier locations (Phoenix, Scottsbluff, Salt Lake City, Seattle, Boston, and San Jose) does not interfere with the requirements of ASHRAE Standard 62R. Furthermore, this ventilation strategy reduces the energy consumption and peak power demand due to air-conditioning the space, as compared to

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the ventilation strategy requiring continuous space ventilation. Consequently, to allow for the optimal design of the two systems from the point of view of their energy consumption, the ventilation of the base-case space should be interrupted at night (the second ventilation strategy should be employed) at the drier locations.

(3) Due to the design of the parametric study, the simulated radiant cooling system and the all-air system cool and dehumidify the same amount of outside air, and provide a similar air relative humidity inside the base-case space. Because the two systems consume the same amount of dehumidification energy, the dehumidification process does not provide any "opportunity for savings" (nor any energy penalties) to the radiant cooling system. Conversely, the sensible load on the cooling coil, and the fan load due to air distribution of the all-air system offer "opportunity for savings" The savings achieved by the radiant cooling system should therefore correlate with the sensible cooling and fan energy consumption (or peak power demand) of the all-air system.

Figure 5.10 presents the energy savings of the radiant cooling system as a function of the sum between the sensible cooling and the fan energy consumption of the all-air system.[1] The data points and the solid-and-dotted line in the figure correspond to the results reported in Section 5.5.1 and 5.5.2, which assume a COP of 3 for the cooling coil-chiller combinations serving the two systems. The dashed lines in Figure 5.10 correspond to similar calculations of the all-air system and radiant cooling system energy consumption, performed with the assumption that the cooling coil-chiller combinations serving both systems have COP values of 2.5 and 6, respectively.

The linear regression between the two quantities indicates that the radiant cooling system can achieve high energy savings at the locations where the sum between the sensible cooling coil and fan energy consumption of the all-air system is high. An examination of the locations associated with the data points in Figure 5.10 shows that the absolute energy savings are highest in the hot climates and lowest in the cold climates, regardless of the dehumidification energy consumption.

The regression line for COP = 3 also shows that, at locations where the sum between the seasonal sensible cooling and fan energy consumption of the all-air system is lower than $10\,\mathrm{kWh_e/m^2}$, replacing the all-air system with a radiant cooling system will not save any energy. The $10\,\mathrm{kWh_e/m^2}$ value can be interpreted as the sensible cooling and fan energy consumption associated with supplying only the ventilation air to the space. Among the locations examined, Seattle presents the lowest sum between the seasonal sensible cooling and fan energy consumption of the all-air system: $18.1\,\mathrm{kWh_e/m^2}$.

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1. The energy savings of the radiant cooling system were calculated as the absolute difference between the total (sensible, latent and distribution) energy consumption of the all-air system and the total energy consumption of the radiant cooling system. To obtain the seasonal energy savings, the seasonal sum between the cooling and fan energy consumption of the all-air system, the calculations were done for the typical week, then extrapolated to the cooling season (see Section 4.4.4).
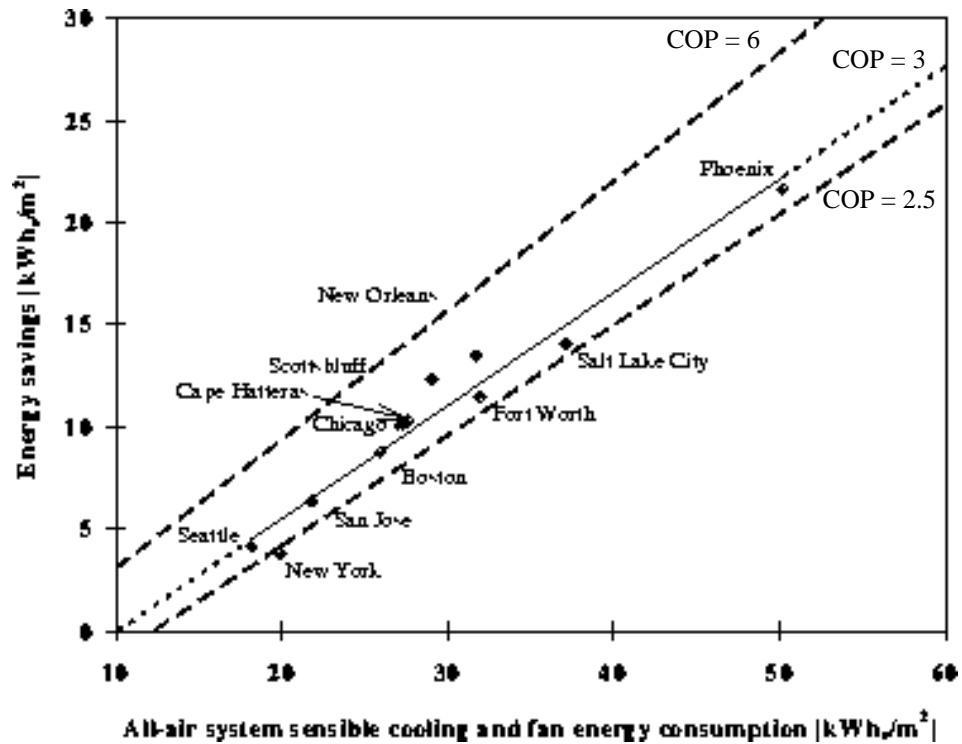
Figure 5.10. Energy savings over the cooling season: trend across climates.

The regression line corresponding to COP = 2.5 has a slightly lower slope than the slope of the regression line for COP = 3, while the regression line corresponding to COP = 6 has a slightly higher slope. Consequently, if the chiller consumes less electrical energy to achieve the same thermal cooling at the coil, the fraction of the sensible cooling and fan energy that can be saved by replacing the simulated all-air system with the simulated radiant cooling system increases. It is important to note that the "closeness" of the regression lines corresponding to different COP values is due to the assumptions embedded in the parametric study. Although it is difficult to estimate the applicability of these results in other situations, the existence of a linear relationship between the savings achieved by the radiant cooling system and the "opportunity for savings" offered by the all-air system is an important result.

Section 5.4.1 presents the energy savings of the radiant cooling system as fractional savings.[1] The solid-and-dotted regression line corresponds to a cooling coil-chiller combi-

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1. The fractional energy savings were calculated as the difference between the total (sensible, latent and distribution) energy consumption of the all-air system and the total energy consumption of the radiant cooling system, divided by the total energy consumption of the all-air system.
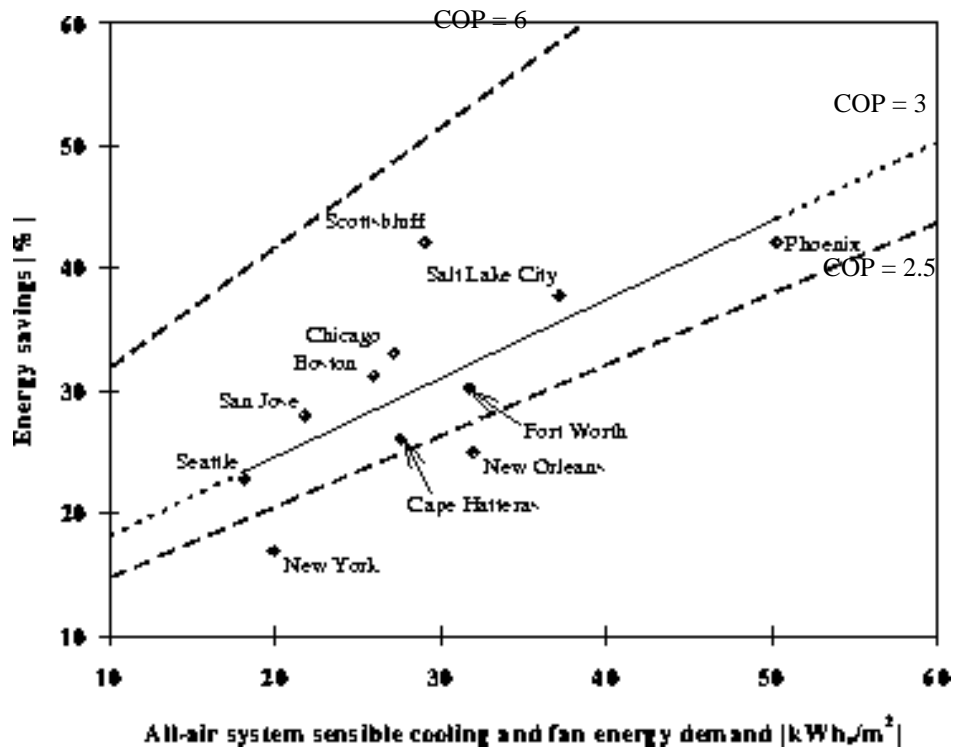
Figure 5.11. Fractional energy savings over the cooling season: trend across climates.

nation with a COP of 3, and the dashed regression lines to cooling coil-chiller combinations with COP values of 2.5 and 6, respectively. Figure 5.11 shows that the correlation between the fractional savings of the radiant cooling system and the sensible cooling and fan energy consumption of the all-air system is not very strong (the data points have a wider spread around the regression line). This result is intuitively correct, because the fractional savings also depend on the latent energy consumption of the cooling coil serving the all-air system terminal.

Figure 5.11 also shows that the fraction of the sensible cooling and fan all-air system energy consumption that can be saved by replacing this system with the radiant cooling system is highest in hot dry climates and lowest in cold moist climates. This result confirms the earlier observations regarding the "opportunity for energy savings" of the radiant cooling system.

The correlations in Figures 5.10 and 5.11 suggest that there is no upper limit for the energy savings that can be achieved by replacing the all-air system with the radiant cooling system. According to Feustel and Stetiu [2], the achievable fractional energy savings

may be as high as 45% for a cooling coil-chiller combination with a COP of 3. On the regression line, this corresponds to a seasonal sum between the sensible cooling and fan energy consumption of the all-air system of roughly 51.5 kW_e/m^2. By comparison, the seasonal sum between the sensible and fan energy consumption of the all-air system is 50.2 kW_e/m^2 at the Phoenix location. Although the Phoenix climate is representative of the hottest US climates, higher values for the energy consumption can be obtained in lighter, less insulated building structures.

Figure 5.12 presents the peak power savings of the simulated radiant cooling system as a function of the sum between the sensible cooling and fan power demand of the simulated all-air system at the time when it reaches its peak demand. The two quantities correlate linearly, indicating that the radiant cooling system can achieve high peak power savings at the locations where the sum between the sensible cooling and fan power demand at the time of the all-air system peak is high. The peak power savings increase with an increase of chiller COP.
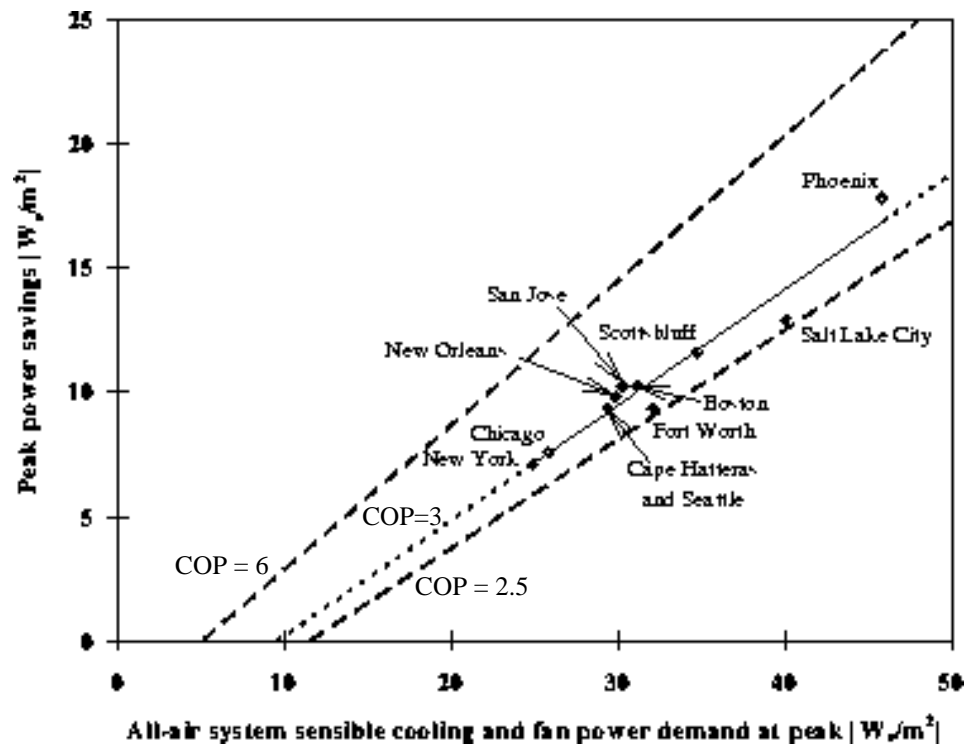

Figure 5.12. Peak power savings: trend across climates.

The data in Figure 5.12 show that the absolute peak power savings are highest in the hot, dry climates, and the lowest in the cold humid climates. Furthermore, the absolute peak

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power savings are relatively high in all dry climates and relatively low in all moist climates. This is intuitively correct, because the "opportunity for savings" at the time of the all-air system peak power demand is high in hot climates, and is low in moist climates.

The regression line for COP = 3 suggests that, if the sum between the cooling and fan power demand of the all-air system at the time of peak is less than 8 W_e/m^2, replacing the all-air system with the radiant cooling system will not save any peak power demand. This value designates the peak sensible cooling and fan load associated with supplying only the fresh air volume to the space.

Figure 5.13 presents the fractional power savings of the radiant cooling system as a function of the sum between the cooling and fan power demand of the all-air system at the time of peak.[1]
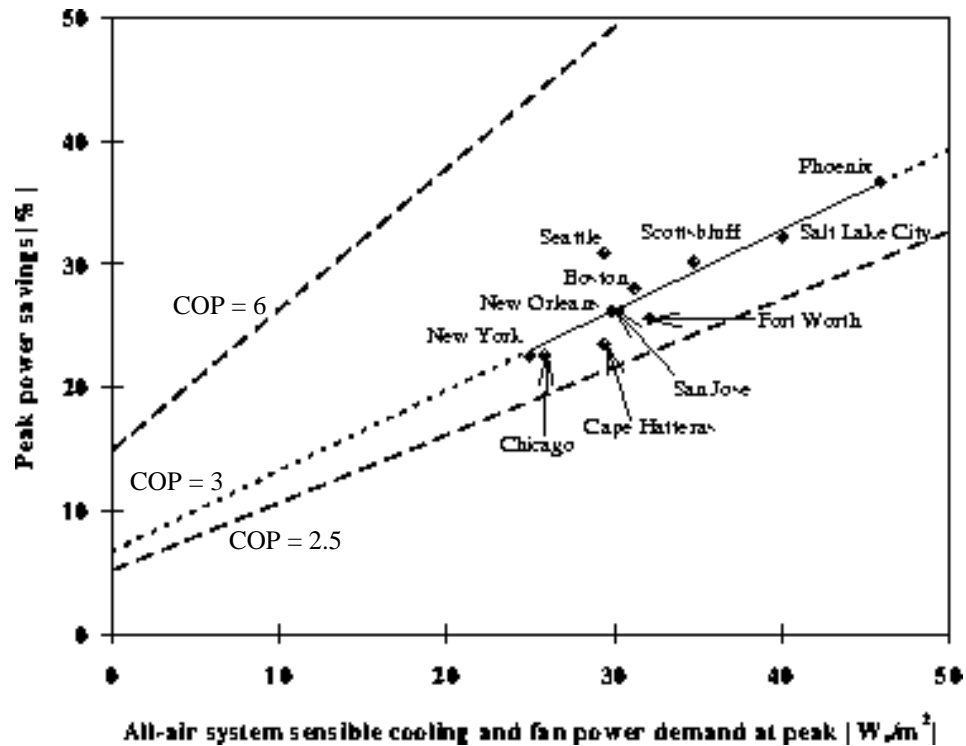

Figure 5.13. Fractional peak power savings: trend across climates.

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1. The fractional power savings were calculated as the difference between the *total* (sensible, latent and fan) peak power demand of the all-air system and the *total* peak power demand of the radiant cooling system, divided by the *total* peak power demand of the all-air system.

As in the case of the fractional energy savings, the correlation for COP = 3 is not as strong as that in Figure 5.12, because the fractional power savings are also a function of the dehumidification load at the time of the system peak. However, Figure 5.13 is consistent with Figure 5.12 in showing that the highest peak power savings are achieved in the hot, dry climates.

The foregoing results can be summarized in the form of a distribution of the energy and peak power savings by number of locations where these savings are achieved (see Figure 5.14). The data used in Figure 5.14 correspond to the ventilation strategy necessary at each location to maintain indoor air relative humidity below the 70% upper limit required by ASHRAE Standard 62R [3]. According to the results in Figure 5.14, replacing the all-air system with the radiant cooling system in the base-case space saves an average of 30% of the energy consumption and 27% of peak power demand of the all-air system conditioning this space.
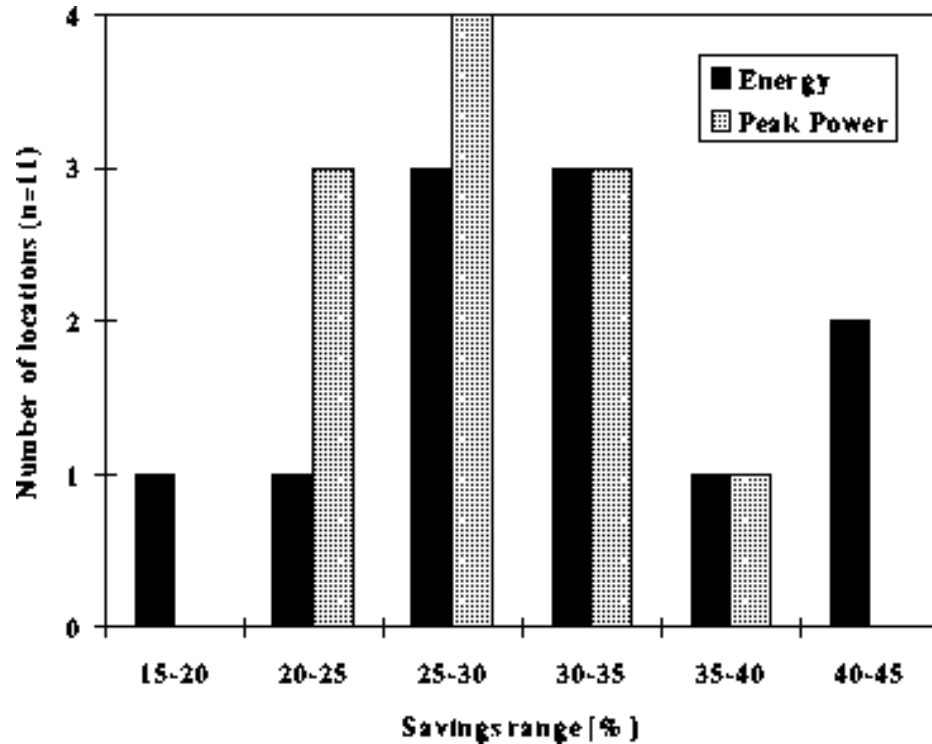

Figure 5.14. Distribution of the energy and peak power savings of the radiant cooling system with the number of locations. Energy average = 30.5% standard deviation = 7.9%. Peak power average = 27.7%, standard deviation = 4.4%.

5.6 Additional Modeling

The results reported in Section 5.5 provide a first estimate for the savings achievable by installing the simulated radiant cooling system instead of the simulated all-air system in a new office space. However, before this information is used to calculate how much energy and peak power *any* radiant cooling system can save if installed in a building, the following limitations have to be considered. First, the results presented in Section 5.5 were obtained by comparing the performance of a simulated radiant cooling system with that of a simulated all-air system that conditions the same specific single-zone office space. It is not certain that the results obtained for the base-case space selected for the study can be used to calculate the savings potential of a radiant cooling system with a different design, conditioning a different space, or a whole building. It is worthwhile mentioning, however, that the integration of RADCOOL into DOE-2 would allow building practitioners to perform simulations for a the building structure of their choice, and to evaluate the savings potential of a radiant cooling system of specified design, as compared to an all-air system of specified design.

Second, the results presented in Section 5.5 were obtained for a new office space with a state-of-the-art envelope complying to current California standards. But the number of new office buildings that will be built in the future is relatively small compared to the number of older office buildings that will be retrofitted. If radiant cooling achieves significant market penetration in the US, radiant cooling systems are more likely to be installed during a retrofit than during the construction of a new structure. It would be interesting to know whether the results obtained for the base-case space in the state-of-the-art structure can be used to draw conclusions about the savings potential of a radiant cooling system in a different building structure.

To extend the building domain where the results obtained in this thesis are applicable, additional modeling is necessary. The following sections will present the results of a few additional simulations. This work explores the extent to which the correlations obtained for the base-case space may change when the energy and peak power savings of the simulated radiant cooling system are calculated for a different space in the building, and for a different building structure.

5.6.1 Description of the additional simulations

To partially address the applicability of the results reported in Section 5.5.3 as an estimate for the energy savings potential of the radiant cooling system in a different space, modeling was performed to simulate the energy consumption and peak power demand associated with conditioning a space with a different orientation. The space MBC6 was chosen for this purpose (see Figure 4.1). The MBC6 space differs from the base-case space MBC2 only through its orientation (north-eastern, as compared to south-western for MBC2). The space MBC6 was simulated in the same conditions as the space MBC2,

but at two locations only: New Orleans and Phoenix. These two locations represent two extreme climates: the New Orleans climate is hot and moist (group 9 in the climate classification described in Section 4.4.3), so the savings potential of the radiant cooling system should be relatively small. By contrast the Phoenix climate is hot and dry (group 3), and the radiant cooling system should achieve high savings. For consistency with the previous work, night ventilation with dehumidified air was simulated in the MBC6 space at the New Orleans location, and no mechanical night ventilation was simulated at the Phoenix location.

To partially address the applicability of the results reported in Section 5.5.3 to calculate the energy savings potential of the radiant cooling system in a different building structure, additional modeling was performed to calculate the energy consumption and peak power demand associated with conditioning the base-case space MBC2 in a building of older vintage. The structure chosen for this purpose has a facade corresponding to the building stock dating from the 1950s: the opaque part consists of metal panels, insulation, and sheetrock, and has a U-value of 1.74 W/m^2-K. The facade has single-pane windows with a center-of-glass U-value of 5.58 W/m^2-K. For simplicity, the interior walls and the ceiling and floor have the same structure as that of the state-of-the-art building. The base-case space was simulated at the same two locations: New Orleans and Phoenix. For consistency with the previous work, night ventilation with dehumidified air was simulated in the MBC6 space at the New Orleans location, and no mechanical night ventilation was simulated at the Phoenix location.

It is important to note that, for consistency with the previous work, the simulation of the space with the "older" building structure was made assuming (1) the same (relatively low) internal loads as those in the parametric study, and (2) the possibility of avoiding infiltration at the New Orleans location by pressurizing the space. Depending on the building to be retrofitted, one or both of these assumptions may not hold. High internal loads at hot dry locations might indicate that radiant cooling systems do not have enough cooling power to condition certain retrofitted buildings. High infiltration rates *and* high internal loads at hot humid locations might indicate a relatively high risk of condensation in certain buildings, even if continuous ventilation is employed. In such extreme conditions, the decision to install a radiant cooling system must be based on simulations performed for each retrofitted building separately. The building practitioner must then make a decision based on (1) the lowest acceptable energy savings of the radiant cooling system as compared to an all-air system, and (2) the highest acceptable risk of condensation.

5.6.2 Results of the additional simulations

Figures 5.15 and 5.16 show graphs similar to those in Figures 5.10 and 5.12. The data points represent the energy savings calculated for (1) the space with south-western orientation, in the "new" building structure (diamonds), (2) the space with north-eastern orientation, in the "new" building structure (triangles), and (3) the space with south-
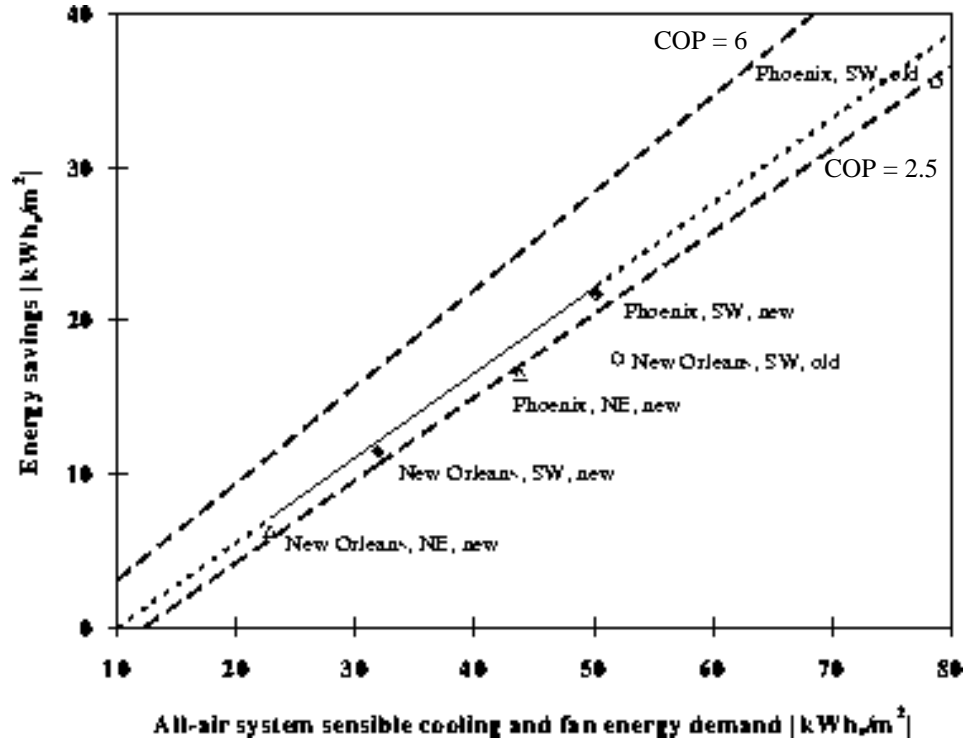
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Figure 5.15. Energy savings over the cooling season: data for New Orleans and Phoenix.

western orientation, in the "old" building structure (circles). The data points correspond to a COP of 3 for the cooling coil-chiller combinations serving both systems. The regression lines in Figures 5.15 and 5.16 are the same as those in Figures 5.10 and 5.12.

To understand the position of the new points relative to the regression line, it is important to remember that the radiant system has more "opportunity for savings" when (1) the sensible cooling load is large, and (2) the all-air system requires a large fan for cooling the space.

The space with north-eastern exposure is exposed to sunshine mainly in the morning hours. Therefore, at the time of the maximum solar heat gain (around 9 a.m.), the building structure has not had a chance to warm up. The relatively cold building surfaces store some of the heat, so the sensible cooling and fan loads imposed on the system are somewhat diminished.

By comparison, the maximum solar heat gain occurs around 3 p.m. in the space with south-western orientation. At this time the building structure is already warm, and the building surfaces can store very little additional heat. Consequently, the system that
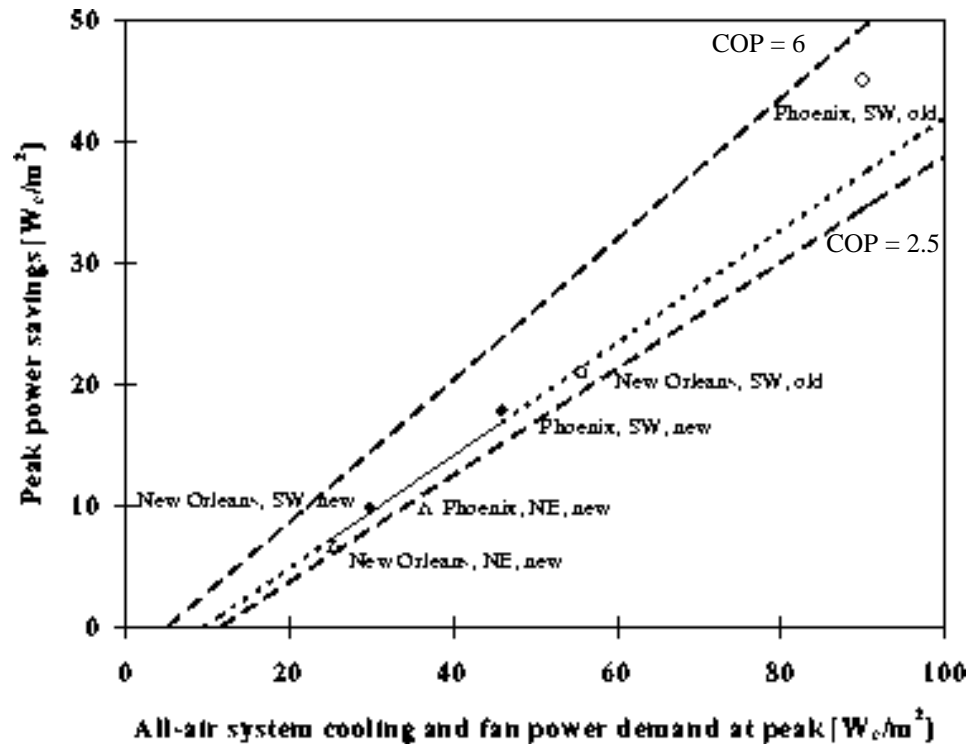
Figure 5.16. Peak power savings: data for New Orleans and Phoenix.

cools the MBC6 space must remove a smaller weather-induced heat load than the system that cools the MBC2 space. The "opportunity for savings" is lower for the MBC6 space than for the MBC2 space, therefore the potential energy and peak power savings are lower. Figures 5.15 and 5.16 confirm this statement.

The solar gain through a poorly insulated structure is larger than the solar gain through a well insulated structure. The results of modeling the "old" structure confirms this observation (Figures 5.15 and 5.16). Because the rate of outside air supplied to the "old" building structure is the same as that supplied to the "new" building structure, the higher savings achieved by the radiant system in the "old" building structure are solely due to the low insulation level of the "old" building. This result indicates that, if the internal loads are not too high and the radiant cooling system can condition buildings of older vintage with a relatively low risk of condensation, the energy and peak power savings achieved by installing radiant cooling systems in retrofit projects be might be larger than those estimated in Section 5.5.

It is important to note that, although the results reported in Sections 5.5 and 5.6 reflect the specific assumptions embedded in the parametric study (occupant and equipment sched-

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ules, design and operation of the all-air and radiant systems, the method of matching the indoor conditions of the space, etc.), they confirm that substantial energy and power savings can be achieved by substituting radiation for convection as a heat transfer mechanism, and water for air as a heat transfer medium. Once RADCOOL integration into DOE-2 is achieved, building practitioners will be able to perform similar studies using any specific assumptions.

5.7 Conclusions

(1) Different ventilation strategies are necessary at different locations to ensure that office building conditions comply with the upcoming building regulation (at least with the revised version of ASHRAE Standard 62-1989 [3]). The design of the ventilation strategy for a building, and the design parameters of the building conditioning system, should therefore reflect local climate characteristics. Specifically, the indoor relative humidity of office buildings located in moist climates should be controlled through continuous ventilation with dehumidified air. Because humidity buildup does not constitute a problem in dry climates, moisture control through night ventilation is not necessary in these climates.

(2) An adequately designed and operated radiant cooling system can function in a state-of-the art office building at any US location with a small risk of condensation. In humid climates, the risk of condensation on the radiant surface is greatly reduced if the building is continuously ventilated with dehumidified outside air. Continuous ventilation may fail to lower the risk of condensation to acceptable levels in leaky buildings of older vintage.

(3) Over a 24-hour period, the simulated indoor air temperature in the base-case space conditioned by the radiant cooling system is more stable than the simulated indoor air temperature in the base-case space conditioned with the all-air system.

(4) The simulated radiant cooling system requires less energy and peak power to condition the base-case space than the simulated all-air system. At the locations studied, and in a state-of-the-art office space conditioned to meet the requirements of ASHRAE Standard 62R, the average savings potential of the simulated radiant cooling system is 30% for the energy consumption, and 27% for the peak power demand. If radiant cooling systems can remove the higher cooling loads characteristic for buildings of older vintage, higher savings are achievable in these lighter structures.

(5) The potential savings of the simulated radiant cooling system are lower in cold, moist climates and higher in hot, dry climates. At the locations studied, the achievable energy savings of the system conditioning the base-case space vary between 17% and 42%. The achievable peak power savings vary between 22% and 37%.

(6) The estimated energy and peak power savings increase when the COP of the cooling coil-chiller combination serving the air-conditioning terminal increases.

(7) If the sum between the seasonal sensible cooling and fan energy consumption of the all-air system drops below the level at which ventilation air is sufficient for cooling and dehumidification, the "opportunity for energy savings" disappears. Replacing the all-air system with a radiant cooling system will not reduce energy consumption. A similar statement can be made for the peak power demand.

(8) Additional modeling is necessary to clarify to what extent the results presented in this thesis are applicable to other building structures and to other orientations. In particular, since retrofit projects will probably account for a large share of the construction projects in the near future, the savings potential of radiant cooling systems in retrofit projects should be studied in detail. Installing a radiant cooling system in retrofit projects should be preceded by simulations reflecting the conditions for each retrofit situation. RADCOOL integration into DOE-2 would provide building practitioners with a simulation tool capable of evaluating the performance of radiant cooling systems in any specific building and for any specific climate.

(9) Because many other alternative cooling technologies are viable in hot, dry climates (e.g. cooling towers, evaporative cooling, night ventilation), it is recommended that pilot-projects demonstrating the performance of radiant cooling systems be implemented in the warm and hot humid climates first. This thesis has shown that installing a radiant cooling system instead of an all-air system in new building construction in these climates can reduce the energy consumption and peak power demand due to air-conditioning by an estimated 25%. Of the existing commercial building stock, about 23% is located in warm and hot humid climates (see Table 4.2).

5.8 References

1. ASHRAE Standard 55-1992, *Thermal environmental conditions for human occupancy*. Atlanta: American Society of Heating Refrigerating and Air-Conditioning Engineers, Inc., 1992.

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3. ASHRAE Standard 62-Revised (to be published), *Ventilation for acceptable indoor air quality*. Atlanta: American Society of Heating Refrigerating and Air-Conditioning Engineers, Inc.